DESIGN OF COLD WATER PIPE
FOR SEA THERMAL POWER PLANTS

Progress Report
for Period 1 May 1975 - 15 September 1975

J. Hilbert Anderson

Sea Solar Power, Inc.
York, Pennsylvania 17403

MASTER

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product or process disclosed or represents that its use would not infringe privately owned rights.

September 1975

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Prepared For

THE U.S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
UNDER CONTRACT NO. E(11-1)-2691

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

ABSTRACT

This report covers the preliminary analysis of design conditions for a 40 ft. diameter cold water supply pipe for a 100 MW sea solar power plant. The 4000 ft. long pipe is assumed to be freely suspended from a floating platform. Internal pressure conditions are calculated for maximum assumed flow rates in the pipe. The design is based on a circular row of tubes with spacers between to form the pipe wall. Collapse stability calculations are made for the pipe and for the individual tubes. Drag and moment distributions are calculated for the pipe, and stress calculations made for these preliminary design conditions. It is concluded that the basic design is feasible and justifies more precise analyses.

TABLE OF CONTENTS

	<u>Page No.</u>
INTRODUCTION	1
TECHNICAL SECTIONS	
A. PIPE REQUIREMENTS	2
B. DESIGN CONCEPTS	6
C. ANALYTICAL PROCEDURE	9
CONCLUSIONS	27
RECOMMENDATIONS	27
REFERENCES	29
APPENDIX	30
Fig. 1. Plant Water Requirements	
Fig. 2. Diagram of Pipe Construction	
Fig. 3. Tube Spacer Arrangement	
Fig. 4. Tube Joint	
Fig. 5. Pipe Support	
Fig. 6. Horizontal Load Diagram	
Fig. 7. Horizontal Load Diagram	
Fig. 8. Vertical Load Diagram	
Fig. 9. Tubes with I Spacers	

INTRODUCTION

A sea thermal power plant requires huge quantities of cold water for condensing purposes, brought up from ocean depths of approximately 2,000 to 5,000 ft. Typically, for a 100 MW plant, volume flows of 8,000 to 25,000 cubic ft. per second might be required. The exact quantity for any particular design plant depends on optimization of all of the other equipment involved, in order to make the most effective economic cycle for power generation, and/or production of fresh water, chemicals, and food. From earlier work, we have chosen a pipe design diameter of 40 ft. as being typical of what might be used to supply cold water to a 100 MW power plant.

Any large cold water pipe suspended from a vessel in the ocean must be attached to the vessel. Since a floating power plant is subject to ocean forces such as waves and winds, it is almost certain to be subjected to forces large enough to move the vessel in the usual modes of heave, yaw, pitch, and roll. Since a suspended pipe up to 4,000 ft. in length becomes a large mass, then any connection between the pipe and the vessel must necessarily allow for the motions and forces between the vessel and the pipe. Early failures by Claude in trying to make a floating power plant with a suspended cold water pipe were failures simply because the cold water pipe broke away from the vessel. This is a most serious problem and must be recognized in any potential cold water pipe design.

This report discusses in detail the various problems that must be solved in connection with the cold water pipe design, and presents some early conclusions as to their solution.

TECHNICAL SECTIONS

A. REQUIREMENTS

The first requirement for any cold water pipe is that it must be strong enough to resist all of the forces involved in its association with transporting the water and the ocean climate surrounding it. The pipe must accept the drag forces produced by the ocean currents surrounding the pipe. These vary over the depth of the pipe, as well as with time and location. The pipe must also be strong enough to resist any forces imposed on it by the platform supporting it. Since the platform is subject to wave forces in the upper levels of the ocean, then the movement of the platform will normally produce variable forces on the pipe, which can be vertical, horizontal, or angular motion.

Since the pipe is carrying water which must be lifted from deep in the ocean, there is necessarily a lower pressure inside the pipe than external to the pipe. This pressure difference is partly due to the conversion of potential energy into kinetic energy of flow, partly due to friction losses, and partly due to the difference in density between the cold water on the inside of the pipe and the warmer surrounding waters. Since the pressure inside the pipe is lower than the pressure surrounding the pipe, then the pipe must be designed so that it will not collapse due to the external pressure difference across the wall. In any pipe that has a small thickness-to-diameter ratio, the unstable collapse problem is a serious one. Even though the pressure difference across the wall in this case is quite small by ordinary standards, it can become a very important problem in the design of this pipe. Any bending stresses on the pipe due to currents also increase the tendency to collapse, and must be taken into account in the pipe structure design.

Ocean currents flowing around a circular pipe create unstable vortices, which generate vibrating forces on the pipe. This is a familiar effect on long electric cable transmission lines, where failures have occurred due to cross winds causing the lines to vibrate. This type of self-excited vibration has not only caused failures in transmission lines, but in a more dramatic case, actually caused the collapse of the long, narrow Tacoma suspension bridge. The design of the cold water pipe must take account of these self-excited vibration forces, and make the pipe resistant to failure from this cause.

Since there will inevitably be some pulsating forces on the pipe due to the self-excited vibration, then the design should be such that it will, as much as possible, damp out the stresses generated by these vibrating forces,

and have fatigue resistance to the application of repetitive forces.

In order to get damping effects and reduce stress concentration problems that might possibly be caused by vibrating effects, or sudden local stress changes, it is extremely desirable to make the pipe flexible in design. By making the pipe flexible, we adopt the principles long established in nature that show that stress problems are reduced by allowing considerable movement in any object, thereby allowing greater damping in the structure as well as reducing the rate of transmission of stresses from one part to another.

The very nature of this system requires that a long pipe be suspended from the floating platform. Since the platform is going to move when wind and wave forces operate on it, and the pipe is a large mass connected to it, then any motion of the platform tends to produce extremely large forces on the pipe mass. For this reason, it appears logical to allow the platform to move at least partially independently of the pipe. We therefore need to design a connection between the pipe and the platform that has the lowest possible force transmission between the two when the platform moves. Ideally, we would like to have a connection which transmits only a constant force, sufficiently large to support the negative buoyancy of the pipe hanging from the platform.

The force required to support the pipe is a function not only of the mass of the pipe, but also of its buoyancy. We cannot make the pipe completely self-buoyant, because it would then not sink to the desired depth. However, by building in a certain amount of buoyancy in the pipe, we can reduce the force required to support it from the structure. Therefore it is desirable to build buoyancy into the pipe itself, rather than have the supporting buoyancy provided by the platform. This will reduce the necessary force in the connection between the two.

The connection between the pipe and the platform must allow for vertical movement of the platform with respect to the pipe; it must allow for angular motion to take care of pitching and rolling of the platform, and it may even be required to allow for some yaw in the platform. The problem, then, is to allow all of these relative motions without producing excessive forces at the connection between the pipe and the platform.

The pipe connection should also, desirably, be in operation as a flexible support during construction of the pipe. Any of the forces mentioned might occur during construction, and it is just as important to protect the connection between pipe and platform during construction as it is following construction.

Another possible force problem in connection with the pipe is the inertia of the flowing water. Since we have a very large volume of water flowing at an appreciable velocity, then any sudden variations in flow will potentially produce stresses on the pipe or on its connection to the vessel. This is a familiar problem in hydroelectric water transmission lines.

Since a large quantity of water must be pumped, and the power required to pump it subtracts directly from the usable power for sale from the plant, then pumping losses must be assessed at a value of the total plant cost to produce the power required to overcome those losses. It becomes important, then, to design a cold water pipe which has low inherent friction losses. The inlet to the pipe must also be carefully designed so that high losses do not occur. For example, a straight pipe end would incur a direct loss of about 40% of the kinetic energy in the pipe at the inlet. The exit losses from the pipe also need to be carefully controlled. The total kinetic energy in the pipe can become the exit loss if the discharge is not carefully designed to recover the kinetic energy as the water leaves the pipe.

It appears advisable to put a screen at the inlet of the cold water pipe. While we cannot hope to design a screen for the cold water inlet that is sufficiently fine to protect the heat exchanger passages, it is still logical to keep out any possible large animals from being sucked into the pipe.

The large size and large quantity of water being handled make heat leakage from the surrounding warm water into the cold water not too much of a problem. However, when you realize that 1°F. will make approximately 5% difference in plant output, then it is certainly desirable to design a pipe which does have a minimum flow of heat into the cold water from the surrounding warm water.

Since the pressure in the pipe is lower than the surrounding water pressure, there is a tendency for leakage of warm water into the pipe as the water flows upward. This possibility must be recognized, and the pipe should be designed to minimize any leaks from the outside. It should also, preferably, be possible to repair leaks if they do occur.

It has been proposed by some to construct the cold water pipe at a shore base and transport it to the ocean platform, then sink it into position. This is pretty obviously a stupendous, and difficult and costly task. We believe it to be far more desirable to be able to transport easily transportable materials to the plant platform, and then to be able to assemble these parts and construct the pipe on the platform at sea. In order to do this, we need a design which can be constructed on the platform at sea, and can be so constructed

that it is lowered from the platform, one section at a time, as the pipe is being constructed. By its very nature, this requires that the supporting structure be usable as construction proceeds as well as after the pipe is finished and ready for service.

As another desirable, although not completely necessary, feature, the pipe should be retrievable from the ocean after long-time service. Since an appreciable amount of material will be used, it would be desirable to be able to retrieve this material after the pipe has possibly deteriorated to the point of requiring replacement.

While it is desirable that the pipe be fouling-resistant, this is probably not a very serious problem. It has already been found from experience at St. Croix that the inside of a pipe carrying cold water from deep in the ocean does not foul appreciably. Since we will probably be dealing with pretty high velocities inside the pipe, then there is very little chance of any fouling occurring. In the upper layers close to the platform, there is some possibility of fouling of the external structure, but this is not likely to be a problem over the greater part of the pipe length.

The problem of corrosion is always with us in the ocean. Therefore the materials used should be chosen to minimize the possibility of corrosion. However, since the water from deep in the ocean is quite cold, there is a little less likelihood of corrosion problems in the cold water pipe than there is in the other parts of the structure. Despite this favorable condition, protection methods should be provided, if at all possible.

Many materials for a cold water pipe have been suggested, ranging from wood, to plastic, to steel, to aluminum, to concrete. All of these materials are potential candidates, and all have their advantages and disadvantages.

The final requirement for the cold water pipe is that it should be low in cost. Since we must handle vast quantities of water over a fairly long distance to produce a relatively small amount of power, then the cost of materials and building the cold water pipe does become an important factor in the total plant cost. However, it is interesting to note from Figure 1 that the amount of water that needs to be handled in a sea thermal plant per unit of power produced is of the same order of magnitude as the amount of power that flows through a hydroelectric plant. Therefore it is not likely that the cold water pipe will be a limiting factor in the cost of sea thermal power.

B. CONCEPTS

After some 10 years of study and investigation, looking for the ideal design of a cold water pipe and making preliminary evaluations of many types of construction, we tentatively decided that the design that would fulfill all of the foregoing requirements with the greatest probability of success is a construction as depicted in Figure 2. This consists of a multiplicity of tubes, arranged in a circle 40 feet in diameter, with interconnecting spacers between the tubes as is tentatively shown in Figure 3. The tubes could be joined to each other and, being hollow, could provide their own buoyancy.

Most of the concepts that have been suggested by different people do satisfy some of the requirements, but, little by little, we have become convinced that this construction is most likely to satisfy all of the requirements.

The multiplicity of hollow tubes provides the flexibility in the system corresponding to each individual tube, thereby making a much more flexible structure than would be the case with a single-wall circular pipe. The multiple tube elements, being clamped to each other, also provide a great amount of damping friction in the structure, so that any vibration forces tend to be damped out in friction of the individual parts on each other. The principle is rather similar to that which has been successfully used to construct silos, which are rather high, circular structures with high wind loading. The hollow wall raises the thickness-to-diameter ratio of the structure so that the unstable collapsing problem is minimized. The high damping factor of individual elements should reduce tremendously the possibility of self-excited vibrations from currents being built up to any dangerous level. This might not be the case with any single-wall structure. The individual hollow tubes create a roughening effect on the exterior, which should develop small, individual eddies as the currents flow past the pipe, and this should tend to minimize very large eddies that cause self-excited transverse vibrations. This effect must finally be analyzed by actual testing.

This type of pipe construction should have a very low friction loss, since the inside of the pipe can be perfectly smooth in the direction of flow. In the construction as conceived, the joints between individual tubes will be purely mechanical, expanded joints, with their external diameter exactly equal to the tube diameter. Therefore there is no obstruction inside in the flow stream.

Since the hollow wall construction is relatively thick, this almost eliminates any measurable heat leakage from the water outside to the cold water inside the pipe.

This pretty well fills the desirable requirement that materials can be transported easily at sea and constructed on the platform. For the most part, all parts are reasonably small, being pieces of ordinary pipe. These can easily be transported by ship to the platform and stored on the platform during the construction period. The hollow tube construction permits fairly accurate buoyancy control. For example, the lower tubes can be filled with water, or even with rock, so as to produce high-density pipe at the bottom, whereas the upper part of the hollow wall can be filled with gas so that it provides a very large share of the total buoyancy required. This means that the support at the platform does not need to carry a great deal of weight, but must allow for the inertia of the entire pipe mass.

By using entirely mechanical joints expanded into the tubes (as shown on Fig. 4), this makes it possible to use coated pipe materials, if so desired. This means that handling at sea is much less of a chore, since no welding is required that would obviously destroy any coating of the pipes that were used in the assembly.

The construction lends itself to the possible use of several different materials. For the tubes, there is a possibility of using either steel, aluminum, or fiber glass plastic. Tubes from any of these materials are already in production, and the construction should lend itself to the use of any of these materials.

The pipe support construction is illustrated diagrammatically in Figure 5. The upward force on the pipe is produced by hydraulic pressure supplied by an external pump, pumping in water from the sea. The water pressure underneath the piston provides an upward force, which supports the negative buoyancy in the pipe. The piston is flexible, being made so by being constructed of a series of multiple plates, each of which can be flexed sufficiently to allow for approximately 5° from the normal vertical position. The piston is loosely sealed into the open well, and probably will have rolling supports at the outside diameter to supply smooth operation up and down. The allowable movement of the piston in the open well must be sufficient to allow for any amount of vertical movement of the platform with respect to the pipe. The bottom of the open well must be sealed against too much leakage of water outward to the open sea. This seal is accomplished by having a loosely fitting seal plate surrounding the pipe, which can slide about in the well, and provide a moderate seal to prevent too much water from flowing out of the pressurized space. Relative movements of the piston upward or downward in the open well require rather sudden changes in the water flow from underneath the piston. By using a centrifugal pump which has a reasonably flat characteristic and having a reasonably large seal leakage corresponding to normal pump flow, then

movements up and down are merely taken by adjustments in the flow over a wide range through the pump. This, in effect, acts as a hydraulic spring with a very low spring stiffness.

In order to provide redundancy in the available hydraulic support pressure, several pumps would be used in parallel, with possibly several different emergency power sources, so that there would be good assurance of always maintaining hydraulic pressure under the piston. In the event of complete power failure in the pumping systems, then the piston needs also to be strong enough to support the pipe, at least temporarily, even though it lands in the bottom of the well.

It is possible to get a wide range of hydraulic spring characteristics by changing the characteristics of the pump, or putting ports of specific shapes at selected locations at the side of the well, so that they tend to balance out the steady state condition at a given operating level for the pipe. Some leakage can also be allowed through the multiple plate diaphragms, and this is rather necessary in order to keep sufficient flow through the pump so that it can easily adjust itself for up and down motions. The net effect of all this is to provide a nearly constant force upward on the pipe corresponding to the negative buoyancy in the pipe. In the event of vertical heave motion of the platform, the forces that are applied to the pipe will change very little within the range of the hydraulic spring rate of the system.

The multi-plate flexible piston diaphragm can also be used during the construction period by making the travel on the piston quite long, so that it can be raised to the top of the well during assembly of the first pipes and then successively lowered by one length of an individual section. This provides a movable platform, which permits successive lowering of pipe sections down into the sea as construction proceeds. By this system, construction can proceed above the deck, section by section, and the hydraulic piston can be used to lower the pipe as it is being constructed. The advantage of the constant force, or hydraulic spring support, is then maintained through the entire construction period, as well as after the pipe is in operation. Preliminary estimates would indicate that approximately an 80 ft. diameter well would suffice for the support of a 40 ft. diameter pipe.

The cold water pump can advantageously be supported on the piston at the top of the pipe as shown in the diagram. This has the advantage that there is no head loss between the pipe outlet and the pump inlet. The discharge from the pump can then be designed as a diffuser, so that the velocity can be decreased, at the same time converting the kinetic energy of the high-velocity water into pressure energy in the well.

If the well is approximately twice the diameter of the cold water pipe, then the velocity is approximately one-fourth that in the cold water pipe, and the kinetic energy of flow is reduced to approximately 1/16 of that in the cold water pipe. This, then, means that the large open well permits easy access from its openings into passages that lead to the condensers and condenser manifolds. With this arrangement, a very, very low friction loss and loss due to turning the water is achieved.

Since the pressure at the pump outlet is necessarily higher than the surrounding ocean pressure in order to force the water through the condensers with their frictional resistance, then the water level in the open well must be above that of the surrounding sea. For this reason, the open well is shown above the normal deck level of the supporting platform. The open well has the additional advantage that if any disturbance shuts off the water, the inertia of the water itself is easily absorbed merely by a level rise in the open well. It has no appreciable effect on the dynamics of the platform itself. It also should produce no additional forces on the platform.

From a preliminary evaluation of the proposed pipe construction concept, we decided that this concept has an excellent chance to meet all of the requirements of the cold water pipe. No other concept that we are aware of comes so near to having the potential to fulfill all of the desirable requirements for such a pipe.

C. ANALYTICAL PROCEDURE

In trying to design any structure with the complex requirements of the cold water pipe, it is difficult to evaluate the most critical design problems and to know exactly where to start. We therefore decided to start with a more or less logical pipe design, then evaluate its compatibility with all design requirements. The information we obtained from this should provide direction toward a more nearly optimized design.

At the start, we assumed a mean pipe diameter of 40 ft. and a length, from support to bottom, of 4000 ft. (1219 M.). This length is probably longer than the average that might be used, but would make the design problem a little more difficult. If the design proved feasible for this depth, then it should also be feasible for shallower depths.

1. Tube Stresses and Buoyancy

As a trial design, we used tubes made of nominal 6" Schedule 20 steel

pipe. The O.D. = 6.625", and wall thickness = 0.188". For grade "B" pipe, mill test pressure is 1200 psi., and tensile strength is 60,000 psi.

Assuming 1" spacers between tubes, then

$$\text{No. of tubes} = \frac{40 \times 12 \pi}{6.625 + 1} = 198 \text{ tubes.}$$

Weight of tube per ft. length = 12.89 lbs. /ft.

Weight of sea water displaced by tube

$$= \frac{6.625^2 \pi \times 64}{4 \times 144} = 15.32 \text{ lbs. /ft.}$$

Therefore, each tube would be buoyant and would produce $15.32 - 12.89 = 2.43$ lbs. buoyancy per ft. of length, if filled with gas or air. This makes it possible to reduce the downward load on the support to a reasonably small percentage of the total weight of the pipe. By filling the tubes with water or rock at the bottom and with air or gas at the top, we can produce negative buoyancy at the bottom, where it is needed, and positive buoyancy near the top.

Either aluminum or fiber glass reinforced plastic tubes would provide greater buoyancy than steel, and will be investigated later.

There are several possible alternatives for filling the tubes to produce controlled buoyancy:

1. The tubes could be entirely filled with air or gas at atmospheric pressure. Then, at 4000 ft. depth, the external pressure on the tube would be

$$\frac{4100 \times 64}{144} = 1822 \text{ psi.}$$

Note: We assume pipe support is 100 ft. below sea level. This would produce a compressive stress of approximately

$$\frac{1822 \times 6.625}{2 \times .188} = 32,100 \text{ psi. on the tube wall.}$$

At this value of external pressure, the tube wall is subject to collapse due to instability of the circular section. Ref. 1 provides a convenient means for evaluation of the critical pressure that might cause buckling of a circular tube.

Using Eq. (6) of Ref. 1 for the tubes, and assuming a tube length of 40 ft. between joints, we have:

$$L/D = 40 \times 12/6.625 = 72.5$$

$$T/D = .188/6.625 = .0283$$

From Fig. 1, this falls into the range of infinitely long cylinders, and Eq. (7) can be used to determine critical buckling pressure.

$$P_{CR} = \frac{ET^3}{4(1 - \mu^2) R^3}$$

$$P_{CR} = \frac{30 \times 10^6 \times .188^3}{4 \times .91 \times 3.312^3} = 1506 \text{ psi.}$$

Since this value is less than the external pressure of 1822 psi., we cannot use these tubes at this depth filled with air at atmospheric pressure.

2. The tubes could be filled with water. In this case, the internal pressure would be approximately equal to the external pressure, and there would be almost no wall stress from pressure difference across the tube wall. This scheme has the obvious disadvantage that the entire weight of the tubes must be carried by the pipe support, and buoyancy corresponding to this weight must be provided in the platform.

3. The tubes could be filled, or partially filled, with a low density, volatile liquid, such as propane. At 40°F, the liquid density is 32.7 lbs./cu. ft. The vapor pressure at 60°F is 93 psig. If the tube were filled with liquid to within 100 ft. of the top, then the internal pressure at the bottom would be

$$P = 93 + \frac{(4000 - 100) 32.7}{144} = 979 \text{ psig.}$$

The pressure difference across the tube wall at the bottom would be 1822 - 979 = 843 psi. Since this is less than the critical buckling pressure, it would be possible to use the tube under these conditions.

By pressurizing the upper part of the tube with a gas such as methane, the internal pressure could be increased so that pressure difference would be acceptably low.

4. The tubes could be partially filled with water and pressurized with gas at the top. For example, if half filled with water, the internal water pressure at the bottom would be $1822/2 = 911$ psi. If gas pressure in the upper half is raised to 450 psi., the wall pressure differential would then be $1822 - 911 - 450 = 461$ psi. at the bottom and at the middle. The internal pressure would be approximately $450 - 44 = 406$ psi. above external pressure at the top of the tube.

5. The tubes could be partially filled with water, with selective gas pressurization in the upper sections. This would create approximately equalized pressures across the tube wall throughout its length.

6. The tubes could be partially filled with high density rocks at the bottom, and selectively pressurized with gas from the bottom to the top. This scheme has the advantage of producing maximum vertical moment with minimum negative buoyancy. The vertical moment is required to compensate for net horizontal current drag moments on the pipe.

From the above considerations, it is clear that depth and water pressures would not prevent us from using 6" Sched. 20 steel pipes for tubes in the pipe wall. Aluminum alloy or reinforced plastic tubes would potentially provide greater buoyancy, but the steel tubes can provide a reasonable amount of buoyancy.

2. Pipe Compression

The next design consideration is the buckling stability of the pipe against differential pressures created by internal water pressure being less than external water pressure surrounding the pipe.

This reduction in internal pressure is caused by several factors:

1. The 40°F water flowing through the pipe is denser than the surrounding warm water. This results in a lower pressure at the pump inlet in order to lift this denser water.

2. The static pressure inside the pipe is lower than pressure outside

because the water is flowing at relatively high velocity. The kinetic energy of the water must be generated by a pressure difference corresponding to the velocity head.

3. The inlet loss to the pipe contributes to the pressure difference.

4. The friction pressure drop in the pipe reduces the pressure, so that the maximum difference between external and internal pressure occurs immediately before the pump inlet at the upper end of the pipe. The internal area of the pipe is approximately calculated as:

$$A = 40^2 \frac{\pi}{4} - \frac{40\pi \times 12}{6.625} \times \frac{6.625^2 \pi}{8 \times 144} = 1229 \text{ sq. ft.}$$

At a nominal flow of 10,000 cfs. (283 M³/sec.), the mean flow velocity

$$V = \frac{10,000}{1229} = 8.13 \text{ ft./sec.}$$

Since we do not know the best or most economic velocity for this size pipe, it is advisable to assume a higher velocity for these first calculations. We will assume 15,000 cfs. as a possible maximum economic flow.

The velocity is then 12.2 fps. and the pressure difference corresponding to the velocity head is:

$$\Delta P = \frac{\rho V^2}{2g} = \frac{64 \times 12.2^2}{64.3 \times 144} = 1.029 \text{ psi.}$$

If we assume an inlet loss of 25% of this velocity head, then

$$\Delta P = 1.25 \times 1.029 = 1.286 \text{ psi.}$$

The pressure difference due to lifting higher density water through the pipe is approximately 0.9 psi.

The Reynolds number for flow through the pipe is found from the equation:

$$Re = \frac{VD}{\nu}$$

Where V = Velocity in fps.

D = Diameter in ft.

ν = Kinematic viscosity = 1.72×10^{-5} ft.²/sec.

$$Re = \frac{12.2 \times 40}{1.72 \times 10^{-5}} = 2.84 \times 10^7$$

Since this is in the high turbulent flow range, we can compute friction loss from the familiar Darcy formula.

$$\Delta P = f \frac{L}{D} \frac{\rho V^2}{2g}$$

A conservatively high roughness value should be .001, with a corresponding value of $f = .02$. Then

$$\Delta P = .02 \times \frac{4000}{40} \times \frac{64 \times 12.2^2}{64.3 \times 144} = 2.08 \text{ psi.}$$

The total value of ΔP near the upper end of the pipe would then be:

$$\Delta P = 1.286 + 0.9 + 2.08 = 4.27 \text{ psi.}$$

The pipe must then withstand this much pressure difference without buckling. At the lower end, near the inlet, it must withstand a pressure difference of 1.286 psi.

Since the wall is hollow, the effective wall thickness to resist buckling is much greater than the tube wall thickness and would be fairly close to the tube diameter.

From Ref. 3, one can determine an approximation of effective wall thickness to be $6.625 \sin 60^\circ = 5.74''$.

Then, using Eq. (7) from Ref. 1, we can get an approximation to the critical buckling pressure of the pipe. This approximation assumes that no stiffening rings are used and that tubes are bonded to spacers tightly enough to prevent rotational slipping.

Since this is not a solid wall, we must adjust the effective modulus of elasticity in the ratio of actual metal wall thickness to assumed hollow wall thickness.

$$\text{Then } P_{CR} = \frac{2 \times .188}{5.74} \times \frac{30 \times 10^6 \times 5.74^3}{4 \times .91 \times 240^3} = 7.39 \text{ psi.}$$

From this, we can assume that the 6.625" tubes would form a wall thick enough to prevent buckling collapse, even though there were no stiffening rings. Note, however, that this assumes sufficient clamping force or shear bonding at the spacers to prevent rotational slipping of the spacers around the tubes.

The compressive stress on the tube wall must also be checked.

$$S = \frac{Pd}{2t} = \frac{4.27 \times 40 \times 12}{2 \times 2 \times .188} = 5450 \text{ psi.}$$

This is acceptable.

3. Tension Loads

The weight of the pipe produces a direct tension load up to the value of force supplied by the support piston. As a minimum, this must equal the total negative buoyancy of the pipe. As a maximum, it will be the maximum force applied to the mass of the pipe when it is being accelerated upward during heave conditions of the platform. As a starting assumption, we can assume the downward force of an individual tube in water.

$$\text{Tube mass} = 12.89 \text{ lbs. /ft.}$$

$$\text{Water displaced} = (6.625^2 - 6.25^2) \frac{\pi \times 64}{4 \times 144} = 1.685 \text{ lbs. /ft.}$$

$$\text{Net weight in water} = (12.89 - 1.685) 4000 = 44,820 \text{ lbs.}$$

$$\text{Cross sectional area} = (6.625^2 - 6.25^2) \pi / 4 = 3.792 \text{ sq. in.}$$

$$\text{Tensile stress} = 44,820 / 3.792 = 11,819 \text{ psi.}$$

This value appears to be reasonable, but it must be kept in mind that bending forces will possibly add to this. Also, extraneous masses such as spacers, clamping rings, and tube connectors will add to the mass and the tensile loading.

From this, we can conclude that it is highly desirable to reduce this load by using the buoyancy of gas filled tubes to reduce the tensile load on the pipe. This load would also be reduced with aluminum or plastic tubes.

4. Drag Forces

Currents will produce horizontal forces on the pipe. These currents will vary with time and depth.

As a first assumption, we can assume drag forces to be like those about a true cylinder. Data for drag forces generated by flows across a cylinder are shown in Ref. 4, p. 7-102.

If we assume a maximum velocity of 5 ft. per sec. or 2.96 knots, then the Reynolds number is:

$$Re = \frac{VD}{\nu} = \frac{5 \times 40}{1.72 \times 10^{-5}} = 11.6 \times 10^6$$

From Fig. 39 on p. 7-102, the probable drag coefficient $C_D = 0.35$.

$$\text{Drag force} = C_D LD \frac{\rho V^2}{2g} = \frac{.35 \times 1 \times 40 \times 64 \times 5^2}{64.3} = 348 \text{ lbs.}$$

This is the drag force per ft. length of pipe. From Ref. 5, the maximum velocity at a depth of 100 meters is likely to be about 140 cm./sec. or 3.9 ft./sec. in the Florida current, and less than half that value at 200 meters depth. It is not likely that currents faster than this will be found in usual locations for sea thermal power plants.

Drag coefficients are likely to be higher for rough circular cylinders, although no test data is available or known to us at these high Reynolds numbers. There is some indication from drag tests on rough cylinders at lower Reynolds numbers that drag coefficients may be as high as 1.0. If this is the case, then the velocity, which would produce a drag force of 348 lbs./ft., would be

$$5 \times \sqrt{\frac{.35}{1.0}} = 2.96 \text{ ft./sec. or } 90 \text{ cm./sec.}$$

As a first attempt at estimating horizontal pipe forces, the various diagrams of loading shown on Fig. 6 and 7 are shown. They show various possible loadings. Not all of them are realistic. We need to know more about actual velocity profiles, as well as actual drag coefficients, before we can determine actual loading diagrams.

In order for the pipe to be in equilibrium with a pivot support, which produces almost no reaction moment, the pipe must slope in the water, with a downward force. This downward force produces an opposing moment that must balance the moment created by the current drag forces. The calculated moments on Figs. 6 and 7 were based on an assumed constant negative buoyancy along the length of the pipe. While the moments shown are probably not correct for an actual situation and must wait for better data before making a final determination, at the same time they do illustrate some possibilities, and probably show higher moments than those that will be reached in the actual cases.

Figure 8 shows a vertical loading diagram that will probably be preferred over a constant negative buoyancy along the length of the pipe. By concentrating the negative buoyancy near the bottom of the pipe and positive buoyancy near the top, a greater compensating moment is produced for a given pipe slope. This is desirable because less total negative buoyancy is required for any limiting pipe slope.

From Figs. 6 and 7, we selected 334×10^6 lb. ft. as the maximum bending moment required for the pipe design. This corresponds to a constant force of 350 lb./ft. along the upper 2000 ft. of pipe, and a constant opposing force distribution of 150 lb./ft. along the lower 2000 ft. These loadings are probably unrealistically severe, but should serve to point out possible limitations in the design.

It is of interest to note how much moment the pipe could withstand as a summation of the moments on all the individual tubes. This would then require no vertical shear loading between tubes.

$$I/C = 5.935 \text{ in.}^3 \text{ for each tube.}$$

Assuming a limiting yield stress of 30,000 psi.

$$M \text{ per tube} = \frac{SI}{C} = 30,000 \times 5.935 = 178,000 \text{ lb. in.}$$

$$\text{For 198 tubes, } M = \frac{178,000}{12} \times 198 = 2.94 \times 10^6 \text{ lb. ft.}$$

This is much less than the design requirement, and it becomes evident that spacers between tubes must transmit vertical shear loads. This can be transmitted either along the spacer, or at an interlocking joint between tube connecting joints.

If we assume that vertical shear loads are transmitted between tubes, then we can assume the pipe to be a beam of circular cross section, with the

equivalent wall thickness equal to the total cross sectional area of the individual tubes divided by the mean circumference of 480". If we assume spacers 1" thick between tubes, then equivalent wall thickness becomes:

$$t = \frac{\pi dt}{d + 1} = \frac{6.625\pi \times .188}{6.625 + 1} = .513''$$

$$\frac{I}{C} = \frac{\pi}{32} \left(\frac{d_o^4 - d_i^4}{d_o} \right) = \frac{\pi}{32} \left(\frac{(480 + .513)^4 - (480 - .513)^4}{480 + .513} \right)$$

$$\frac{I}{C} = 92,730 \text{ in.}^3$$

$$S = \frac{MC}{I} = \frac{334 \times 10^6 \times 12}{92,730} = 43,220 \text{ psi.}$$

This stress is too high, and we must either use heavier walls or assume a lower required moment. Since the loading diagram used is almost certainly unrealistic, we will assume a somewhat lighter load condition. The Type D loading diagram is probably more realistic, while still being conservative. Choosing the worst case at $u = 350 \text{ lbs./ft.}$, $w = 0$, and $a = 2000 \text{ ft.}$, then $M = 219 \times 10^6 \text{ lb. ft.}$ For this moment

$$S = \frac{219 \times 10^6 \times 12}{92,730} = 28,340 \text{ psi.}$$

This is still too high a stress value, and we must therefore choose heavier wall tubes. The next suitable standard wall thickness is .280". Equivalent thickness is:

$$t = \frac{6.625\pi \times .28}{6.625 + 1} = .764''$$

$$\frac{I}{C} = \frac{\pi}{32} \left[\frac{(480 + .764)^4 - (480 - .764)^4}{480 + .764} \right] = 138,000 \text{ in.}^3$$

$$S = \frac{219 \times 10^6 \times 12}{138,000} = 19,039 \text{ psi.}$$

While this stress might be acceptable for a preliminary design, we have already shown the 6" sched. 20 pipe to be satisfactory in many other ways, such as collapse strength and buoyancy. If we use 6" tubes with .28" wall, we

have less buoyancy control. Therefore at this stage, it becomes logical to use larger tubes, since they will have more buoyancy for the same wall thickness. They also permit the use of a smaller number of tubes, thereby reducing the total number of parts.

If we next choose 8" Sched. 30 pipe as the tube size, the O.D. = 8.625" and wall = .277". Equivalent thickness is:

$$t = \frac{8.625\pi \times .277}{8.625 + 1} = .780"$$

$$\frac{I}{C} = \frac{\pi}{32} \left[\frac{(480 + .78)^4 - (480 - .78)^4}{480 + .78} \right] = 140,917 \text{ in.}^3$$

$$S = \frac{219 \times 10^6 \times 12}{140,917} = 18,649 \text{ psi.}$$

This may be satisfactory if design moment becomes slightly less, as will probably be the case. However, since .279" wall can be obtained in 10" nominal pipe, it may be more satisfactory.

Equivalent thickness is:

$$t = \frac{10.75\pi \times .279}{10.75 + 1} = .802"$$

$$\frac{I}{C} = \frac{\pi}{32} \left[\frac{(480 + .802)^4 - (480 - .802)^4}{480 + .802} \right] = 144,884 \text{ in.}^3$$

$$S = \frac{219 \times 10^6 \times 12}{144,884} = 18,138 \text{ psi.}$$

It may be advisable to increase tube diameter to 12.75" with .281" wall, but we will continue the design on the basis of 10.75" O.D. x .279" wall.

The number of tubes required, with 1" spacers, is now reduced to:

$$n = \frac{480\pi}{10.75 + 1} = 128 \text{ tubes.}$$

It is interesting to evaluate the bending moment that might be resisted by 128 tubes of this size.

$$\frac{I}{C} = 23.42 \text{ in.}^3$$

$$M \text{ per tube} = \frac{SI}{C} = 30,000 \times 23.42 = 702,600 \text{ lb. in.}$$

$$\text{For 128 tubes, } M = \frac{702,600}{12} \times 128 = 7.49 \times 10^6 \text{ lb. ft.}$$

While this is very much higher than that sustainable by the 6" tubes, it is still far below the required value.

5. Shear Stresses

The maximum horizontal shear load is most likely to occur at the top of the pipe, near the support piston. A load diagram of Type D, with $u = 350 \text{ lbs./ft.}$ at the top, $a = 4000$, and $W = 0$ at the bottom, would appear to be a logical maximum. The total shear load at the top would then be:

$$V = \frac{350 \times 4000}{2} = 700,000 \text{ lbs.}$$

$$\text{Shear stress} = \frac{V}{A} = \frac{700,000}{128 \times 9.18} = 595 \text{ psi.}$$

The maximum vertical shear stress in a hollow tube is given by the equation:

$$S_V = \frac{2V}{A} = 2 \times 595 = 1190 \text{ psi. (Ref. 6)}$$

This would be the case for a constant thickness pipe, but since our pipe has a minimum wall thickness of $2t = 2 \times .279 = .558"$, and the mean pipe wall thickness is $.802"$, then the maximum shear stress will be

$$S_V = \frac{1190 \times .802}{.558} = 1710 \text{ psi.}$$

This vertical shear stress can be translated into the maximum vertical shear force per unit length of pipe.

Let V = shear force per ft. of pipe.

$$\text{Then } V = S_V \times 2t \times 12 = 1710 \times .558 \times 12 = 11,452 \text{ lbs./ft.}$$

6. Pipe Clamping

In order to resist the vertical shear force between tubes, we can possibly clamp the tubes tightly enough against the spacers so that friction can resist the shear force.

Assuming that coefficient of friction, $f = .25$

$$\text{Then tension rod force} = \frac{V}{f} = \frac{11,452}{.25} = 45,808 \text{ lbs.}$$

The cross sectional area of the tension rod at 20,000 psi. is:

$$A = \frac{45,808}{20,000} = 2.29 \text{ sq. in.}$$

This would mean that one tension rod of this area would be required in each foot of pipe length. This is obviously not very desirable, and we would therefore be well-advised to design interlocking shear joints as a part of the tube connector. However, if further tests show that shear forces are much less than we have assumed, then the frictional clamping may prove to be feasible.

Note that near the top of the pipe, there is a suction of up to 4.27 psi. This produces an inward force, just as the clamping bands would do. In each wall, the force per ft. of length would be:

$$F = \frac{480'' \times 12'' \times 4.27 \text{ lbs./sq. in.}}{2} = 12,300 \text{ lbs.}$$

This much clamping force is furnished by the pressure, independently of the tension bands.

If it turns out that the total shearing force is less than what we have assumed, then there is an excellent possibility to reduce clamping force requirements, and it is quite possible that independent shear joints will not be required over much of the pipe length. This would have the advantage of increasing frictional damping between the tubes.

It is desirable to maintain compression force between the tubes and the spacers at all times. While the suction pressure inside the pipe automatically maintains this compressive force, this would only act when the cold water pump is operating. Therefore it is important to have tension bands to keep compressive force on the tubes, both when the pump is operating and when it is shut down.

The pipe, when compressed by the tension bands, deflects, becoming smaller at a rate determined by the load vs. deflection rate, or by the spring rate. The tension band also has a spring rate, and stretches as it is tightened on the pipe. When pump suction produces a compressive force on the pipe, it tends to reduce its diameter, and therefore to loosen the tension bands. Because of this effect, it is important to have a greater stiffness rate or spring rate in the pipe than in the tension bands. For this reason, we found that the idea of using rubber spacers between tubes in compression, or thin walled tubes as illustrated on Fig. 3, is not practical. The compressive spring rate would be much too low, and it would be difficult to reduce the spring rate of the tension band to a lower value than that of the pipe.

This situation is similar to that of a wood stave silo, or a wooden barrel. When there is moisture in the staves, they expand and are tightly compressed in the steel tension bands. When they dry out, the staves contract, but since the band stiffness or spring rate is much higher than that of the staves, the bands do not contract with the staves, and they become loose. This then causes the silo to lean, or to collapse.

For the foregoing reasons, we found that a stiffer spacer piece between tubes would be important to keep the compressive spring rate of the pipe high. This resulted in the selection of the construction shown on Fig. 9, with an "I" shaped spacer. The spring rate for this construction can be calculated with reasonable accuracy from Ref. 3, p. 173, Case 8. By using a value of $\theta = 55^\circ$ to 60° , it turns out that the spring rate can be raised to quite a high value, nearly to that of a solid wall pipe.

The construction shown on Fig. 9 provides 3 lines of sealing to minimize leakage into the pipe. Calculations of potential leakage of surrounding warm water into the pipe show that this should be almost insignificant. The important factor is to provide tension bands of sufficiently lower spring rate than the pipe, so that they will maintain tightness under all conditions.

An alternative possible construction is to use extruded pipe with an integral formed spacer as a part of the pipe wall. It may be possible to manufacture such a design, particularly if aluminum is used, but we have not yet investigated this possibility.

A preliminary study of tension bands has been made, and it does indeed appear possible to design a band with the desired characteristics. This will be discussed in a later report.

7. Heat Leaks

The cold water, at 40°F, must pass through upper layers of warm water in its travel to the condenser. If the water temperature rises by as little as one degree, it reduces gross plant output by more than 5% and net plant output by an even greater amount, maybe up to 7%. Therefore it is important to minimize heat leaks through the pipe wall.

As a crude approximation, we can assume that we have 3000 ft. of pipe immersed in water at 70°F. We can also assume a flow of 10,000 cfs., with a velocity of 8.3 ft. per sec.

The mass flow rate of water:

$$W = 10,000 \times 64 \times 3600 = 2.3 \times 10^9 \text{ lbs. /hr.}$$

The area of pipe is:

$$A = 40\pi \times 3000 = 377,000 \text{ sq. ft.}$$

$$\Delta t = 70 - 40 = 30 \text{ F}$$

Heat flow through wall is:

$$Q = \frac{\Delta t \times A}{R}$$

where R is the heat transfer resistance through the wall and surface films.

$$R = \frac{1}{h_o} + \frac{t}{k} + \frac{1}{h_i}$$

$h_o = 100 \text{ Btu/sq. ft. hr. F.}$ for outer wall coefficient, assuming velocity = .35 ft./sec.

$h_i = 1300 \text{ Btu/sq. ft. hr. F.}$ for inner wall coefficient, assuming velocity = 8.3 ft./sec.

$t = 6''$, assumed thickness of pipe wall

$k = 0.33 \text{ Btu in./sq. ft. hr. F.}$, assuming water filled wall.

$$\text{Then } R = \frac{1}{100} + \frac{6}{.33} + \frac{1}{1300} = 18.19$$

$$\text{Then } Q = \frac{30 \times 377,000}{18.19} = 622,000 \text{ Btu/hr.}$$

$$\text{Heat input per lb. of water} = \frac{Q}{W} = \frac{622,000}{2.3 \times 10^9} = .00027 \text{ Btu.}$$

$$\text{Temperature rise in water} = \frac{Q}{W \times \text{spec. ht.}} = \frac{.00027}{.95} = .00028^\circ \text{F.}$$

This is so small that it does not appear worthwhile to refine the calculations further. It should be noted that the upper part of the pipe wall is likely to be filled with gas instead of water. Since the conductivity of gas is far lower than that of water, the heat leak and water temperature rise should be insignificantly small.

As a comparison to the above, we can calculate the heat leak for a single wall steel pipe. If we assume 1" wall thickness with ring stiffeners, then the equation for resistance becomes

$$R = \frac{1}{100} + \frac{1}{314} + \frac{1}{1300} = .0139$$

$$Q = \frac{30 \times 377,000}{.0139} = 811 \times 10^6 \text{ Btu/hr.}$$

$$\text{Heat input per lb. water} = \frac{811 \times 10^6}{2.3 \times 10^9} = .352 \text{ Btu/lb.}$$

$$\text{Temperature rise} = \frac{.352}{.95} = .371^\circ \text{F.}$$

While we have probably been too conservative in assuming a very high mean value of Δt , it is apparent that a single wall pipe can have an appreciable heat leak and temperature rise. The tubular wall pipe has a definite advantage.

8. Buoyancy

The use of positive buoyancy in the pipe reduces the amount of buoyancy that must be built into the platform and reduces the supporting force required

between platform and pipe. Both factors are important.

Buoyancy = Mass of water displaced - mass of pipe.

The following table shows a crude estimate of weights and buoyancy for the 4000 ft. pipe:

	<u>Mass/ft.</u>	<u>Water displacement per ft.</u>
Tubes	4274	5521
Spacers	2332	305
Tube connectors	342	0
Tension bands	600	78
TOTAL	<u>7548 lbs. /ft.</u>	<u>5904 lbs. /ft.</u>
Total for 4000 ft.	30,192,000 lbs.	23,616,000 lbs.

The net force downward on the platform would be 6,576,000 lbs. If the pipe were made of aluminum, the total pipe mass would be reduced to approximately 10,669,000 lbs. The force on the platform could then be reduced to zero, or nearly so, by partially filling the tubes with water. It is obviously desirable to use aluminum or some lighter material, such as plastic impregnated fiber glass tubes. However, it is also possible that thinner wall steel tubes could be used over the lower stressed lengths of the pipe. This should be investigated further and will relate closely to a more accurate determination of drag forces and their distribution.

9. Floating Piston

If we assume that the floating piston must support a negative buoyancy of 6,576,000 lbs. as computed above for the steel pipe, then we can compute water pressure under the piston, based on the piston area.

A logical first assumption for piston diameter is 80 ft. At a diameter ratio of 2 to 1 for piston to pipe, it should be possible to design for 5 degrees angular deflection. Based on 80 ft. diameter, the net piston area is 3770 sq. ft. The differential pressure across the piston to support the load of 6,576,000 lbs. must then be 12.1 psi. This is a very reasonable pressure difference. From it, we can get some idea of the pumping requirements. By arbitrarily assuming a desired rate of movement of one ft. per minute, the volume displacement rate of water is

$$Q = 3770 \times 1 = 3770 \text{ cu. ft. /min.}$$

At 80% efficiency, pumping power would be

$$HP = \frac{3770 \times 12.1 \times 144}{.80 \times 33,000} = 249 \text{ HP.}$$

It may be that higher rates of movement of the piston are desired during construction. This would require more pumps and more power.

To allow for relative vertical motion of the platform with respect to the pipe, the velocity may be higher than that required for raising or lowering the pipe. For example, at a wave period of 16 seconds, and a heave double amplitude of 7 ft., the necessary flow displacement would be

$$3770 \times \frac{2 \times 7}{16} = 3300 \text{ cu. ft. per sec.}$$

This flow must be handled by suitable valves in the piston. It is probable that the hydraulic damping of the system provided by these valves can keep the forces between pipe and platform to an acceptable level and also reduce heave motion of the platform to a value appreciably lower than what would occur without this damping.

CONCLUSIONS

From the foregoing preliminary design analyses, it appears that it would be quite feasible to use the tubular wall construction for the cold water pipe. It is probable that preliminary assumptions have been too severe and that further data and analyses should permit a lighter weight wall to be used. In particular, it would be desirable to use aluminum or fiber glass tubes to make the pipe lighter and have more buoyancy, as well as permit better distribution of buoyancy.

RECOMMENDATIONS

This preliminary study has provided sufficient information to guide us toward what the next steps should be. The following recommendations appear to be important steps toward optimizing a final pipe and support design:

1. We should get measured drag coefficients of fluted structures like this at suitably high Reynolds numbers.
2. We should get motion data on a floating platform in a suitable wave tank, with a simulated cold water pipe suspended from it.
3. We should collect data on ocean currents vs. depth at different typical locations so as to evaluate drag forces much more closely. This has a large effect on design requirements.
4. We should investigate the possibility of using aluminum or fiber glass plastic tubes instead of steel.
5. We should take more detailed evaluation steps of current and drag information that is presently available. The assumptions used for this preliminary design are probably excessively high, but have served to outline the design problems.

6. We need to combine buoyancy and drag effects in various possible combinations.
7. We must go into more detail on piston design.
8. We must develop more detailed analysis of tube joint design, vertical shear analysis, and tension band design.
9. We must investigate pipe angle from vertical, caused by buoyancy vs. drag conditions.

While items 1 through 3 above are beyond the scope of the present investigation, it seems apparent that the preliminary results of this study provide enough encouragement to justify such further investigations.

References

1. "An Analysis of Thin Shelled Cylindrical Bodies," by Albert J. Faulstich, Jr., Design News, October 27, 1969.
2. "Flow of Fluids," Crane Co. Paper No. 410.
3. "Formulas for Stress and Strain," by Roark; McGraw-Hill, p. 173.
4. "Handbook of Engineering Fundamentals," by Eshbach; John Wiley, p. 7 - 102.
5. "Heat Transport of the Florida Current," by Chew, Frank, Wagner, and Work; University of Miami Technical Report No. 57-21, 1957.
6. "Handbook for Mechanical Engineers," by Marks; McGraw-Hill, p. 5 - 23.
7. "Dynacat-semi-submersible for 4000 Ft. of Water," by Bernard Cintract, Ocean Industry, April, 1975, pp. 101 - 107.

FIG. 1

TYPICAL PIPE SEPARATOR

8" SCHED
20 PIPE

THIN WALL
TUBE

FIG. 3

FIG. 4

SKETCH OF SSP COLD PIPE SUPPORT
FIG. 5

TYPE A

$$U = 350 \text{ LB/FT} \quad (5 \text{ FT/ SEC})$$

$$M = 0 \text{ LB-FT} \quad (\text{MAX BENDING MOM})$$

TYPE B

$$U = 350 \text{ LB/FT} \quad (5 \text{ FT/ SEC})$$

$$W = 150 \quad (3.27 \text{ FT/ SEC})$$

$$a = b = 2000 \text{ FT}$$

$$M = 334 \times 10^6 \text{ LB-FT} \quad (\text{MAX BENDING MOM})$$

$$U = 350 \text{ LB/FT} \quad (5 \text{ FT/ SEC})$$

$$W = 0$$

$$a = b = 2000 \text{ FT}$$

$$M = 234 \times 10^6 \text{ LB-FT} \quad (\text{MAX BENDING MOM})$$

FIG. 6

TYPE C

$$\begin{aligned} U &= 350 \text{ LB/FT} \\ W &= 150 \text{ LB/FT} \\ a &= b = 2000 \text{ FT} \\ M &= 191 \times 10^6 \text{ LB-FT} \end{aligned}$$

$$\begin{aligned} U &= 350 \\ W &= 0 \\ a &= b = 2000 \\ M &= 141 \times 10^6 \end{aligned}$$

$$\begin{aligned} U &= 350 \text{ LB/FT} \\ W &= 150 \text{ LB/FT} \\ a &= 3000 \text{ FT} \\ b &= 1000 \text{ FT} \\ M &= 224 \times 10^6 \text{ LB-FT} \end{aligned}$$

$$\begin{aligned} U &= 350 \\ W &= 150 \\ a &= 1000 \\ b &= 3000 \\ M &= 78 \times 10^6 \end{aligned}$$

M = MAX BENDING MOM

TYPE D

$$\begin{aligned} U &= 350 \text{ LB/FT} \\ W &= 350 \text{ LB/FT} \\ a &= 1333 \text{ FT} \\ b &= 2667 \text{ FT} \\ M &= 136 \times 10^6 \text{ LB-FT} \end{aligned}$$

$$\begin{aligned} U &= 350 \\ W &= 0 \\ a &= b = 2000 \\ M &= 219 \times 10^6 \end{aligned}$$

$$\begin{aligned} U &= 350 \\ W &= 0 \\ a &= 1000 \\ b &= 3000 \\ M &= 37 \times 10^6 \end{aligned}$$

$$\begin{aligned} U &= 350 \\ W &= 0 \\ a &= 3000 \\ b &= 1000 \\ M &= 107 \times 10^6 \end{aligned}$$

$$\begin{aligned} U &= 350 \\ W &= 150 \\ a &= b = 2000 \\ M &= 150 \times 10^6 \end{aligned}$$

$$\begin{aligned} U &= 40 \text{ (1 KNOT = 1.7 FT/SEC)} \\ W &= 0 \\ a &= b = 2000 \\ M &= 11.7 \times 10^6 \end{aligned}$$

FIG. 7

VERTICAL LOADING DIAGRAM

FIG. 8

TUBES WITH I SPACERS
FIG. 9